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Design Considerations for Improved Fluidic Input Servovalve Performance

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U.S. Army Electronics Research and Development Command
Harry Diamond Laboratories
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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

A study has been made to determine how the performance of two-stage fluidic input servovalves can be improved. The first stage of the servovalve consists of a fluidic amplifier that is coupled to a set of input bellows, which in turn is coupled to a flapper nozzle valve, and the second stage consists of a sliding spool. At present, fluidic input servovalves have time constants of approximately 15 ms. This study is primarily concerned with determining the first-stage design changes needed to (1) obtain overall fluidic input servovalve time constants of 1 to 5 ms and (2) reduce leakage flow.

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Analysis of the first-stage flapper indicates that the servovalve time constant can be minimized by minimizing the area of the input bellows. The bellows area is the only first-stage servovalve parameter that can be changed to decrease the servovalve time constant without necessitating additional parameter changes. Moreover, minimizing the bellows area also reduces the leakage flow in the servovalve.

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FIGURES

1. INTRODUCTION

The control or power element in many hydraulic feedback control systems is the servovalve. The servovalve varies the rate and the direction of flow of fluid to a fluid motor or an actuator by metering the hydraulic fluid through controlled orifices.[1] A large number of servovalves are electrohydraulic. This type is widely used because electrical devices are ideal for sensing, signal amplification, and computation. On the other hand, the power output and the compactness of hydraulic actuators make them ideally suited as power devices. Thus, the electrohydraulic servovalve serves as an interface as well as a power element in control systems. It converts low-power electrical signals into motion of a valve, which in turn controls large flows or pressures to a hydraulic actuator.[2]

The two-stage electrohydraulic valve (fig. 1) has wide usage and is of primary interest in this study. The two-stage servovalve usually has a nozzle flapper valve for the first or primary stage. The flapper valve is used with the torque motor (fig. 1) to provide a hydraulic pressure or force to move the second- or power-stage spool. The combined torque motor-flapper valve is frequently called a hydraulic amplifier. This type of hydraulic amplifier is well suited for use as a first stage because it has an extremely lightweight moving element (the flapper), which requires very small magnetic forces, thus minimizing the electrical input power required for any given response characteristic. It has comparatively high leakage flow, but since the first stage need not be large, its flow consumption may be held to less than 10 percent of the total flow across the power spool.

The second or power stage in the servovalve in figure 1 employs a spool or a sliding element that moves in a direction perpendicular to the static pressure force or the flow of fluid. It meters the flow of high-pressure fluid to the actuator. This sliding valve has relatively little leakage flow and can be built with very high power gains. Servovalves with the torque motor-flapper nozzle first-stage, spool valve second-stage arrangement can be built with up to 11 kW (15 hp) in capacity with inputs of as little as 10 mW into the torque motor and with outputs of up to 0.001 m^3/s (20 g/m) and time constants of 3 to 5 ms. A two-stage valve of this type is practically insensitive to accelerations and vibrations because the forces available to drive the spool are many times greater than the weight of the spool itself. However, the "stiction" force (the force required to break the spool loose and get it moving) is high with the sliding spool type of second stage, and it can easily be jammed by dirt and impurities.[1]

The invention and the development of flueric amplifiers and flueric or fluidic control elements that can sense, amplify, and compute make it possible to build pneumatic-hydraulic or all-hydraulic control systems. However, the servovalve in such systems must be designed for fluidic, rather than electrical, input signals. Pneumatic-hydraulic and all-hydraulic control systems are of interest because

a. They have the potential for high-frequency response (time constants of 1 to 5 ms).

b. They may be more reliable than conventional systems since they are more rugged.[3]

[1] *A. C. Morse, Electrohydraulic Servomechanisms, McGraw-Hill Book Co., New York (1963).*

[2] *H. E. Merritt, Hydraulic Control Systems, John Wiley and Sons, Inc., New York (1967).*

[3] *R. V. Burton, Design Study—Fluidic Armament Control System (FACS), Honeywell, Inc., Minneapolis, MN, TR-69-2440 (1969).*

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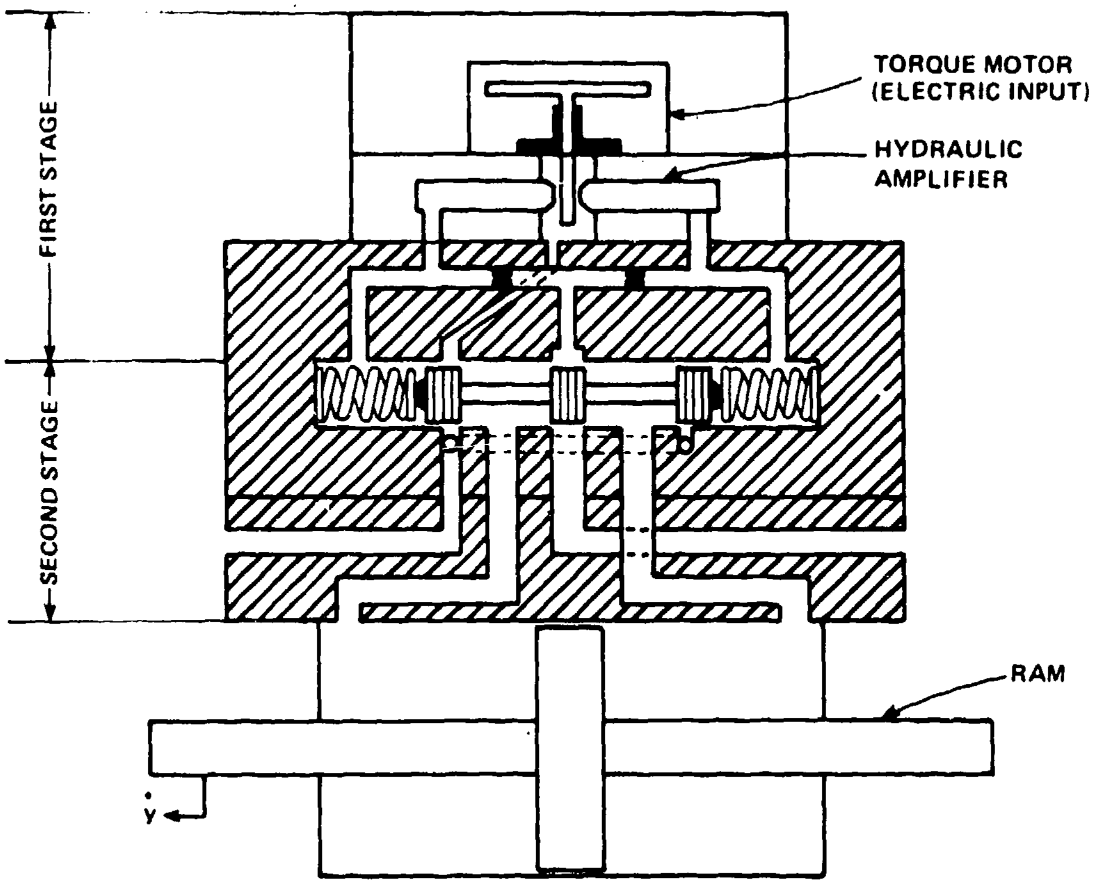

Figure 1. Two-stage electrohydraulic servovalve.

c. They will eliminate electrical-to-hydraulic, mechanical-to-hydraulic, etc., interface devices for all-hydraulic systems.

d. They can be powered by the existing hydraulic transmission power supply.

e. They can possibly be produced and operated at lower cost (few moving parts and no auxiliary power supply for electrical or mechanical components).[4]

At present, fluidic input servovalves have time constants of approximately 15 ms. This study is primarily concerned with the design changes needed to obtain time constants of 1 to 5 ms for fluidic input servovalves and to reduce leakage flow.

[4] *L. R. Kelly and W. H. Booth, Hydraulic Fluidics, American Society of Mechanical Engineers (1968).*

2. DESIGN CONSIDERATIONS

One or more of the following reasons are usually given for using fluidic elements in servovalve design: (1) increasing reliability, (2) lowering production cost, and (3) providing for fluidic input.

There are many possible fluidic servovalve designs. One design[5] calls for an inverted flapper nozzle first or input stage and a vortex valve second stage. A second design[6] uses a fluidic power amplifier (of one or more stages) as a first stage to drive a second-stage spool. A third design calls for the use of a fluidic amplifier cascade and a "jet pipe" to drive the second-stage spool.[7]

A fourth design calls for eliminating the torque motor from the two-stage valve (fig. 1) and attaching mechanical bellows to the flapper arm (fig. 2). The bellows can then be driven by a fluidic amplifier. The operations of the two-stage electrohydraulic servovalve (fig. 1) and the fluidic input servovalve (fig. 2) are basically the same. They differ only in that the input torque applied to the flapper by the torque motor induced magnetic forces in the electrohydraulic version is provided by the fluidic amplifier output pressure used to alternately charge and discharge the bellows in the fluidic version. This fourth servovalve design seems the most promising in terms of minimizing the servovalve time constant and the leakage flow. This two-stage fluidic servovalve arrangement was studied because of its promise and its similarity to the conventional two-stage electrohydraulic servovalve (fig. 1). The design changes needed to minimize the servovalve time constant and the leakage flow in this arrangement are derived in the following sections.

2.1 Derivation of Servovalve Transfer Function

This study is specifically concerned with "simple" first-stage design changes that will minimize the servovalve time constant and the leakage flow. Attention is directed to the first stage of the valve because simple first-stage design changes can probably be made without necessitating redesign of other portions of the valve. Therefore, the transfer function for the fluidic input servovalve (fig. 3) is derived to determine the design parameters that can be adjusted to increase the servovalve frequency response. Neglecting fluidic amplifier input dynamics and transport delay, a pressure difference between the amplifier outputs and the bellows results in a flow between the amplifier and the bellows given by equation (1):

$$P_{A1} - P_{B1} = (L_A s + R_A)Q_1 \, . \tag{1}$$

where

P_A = amplifier output pressure (Pa),

P_B = bellows pressure (Pa),

[5] T. S. Honda and P. S. Ralbousky, *Fluidic Vortex Valve Servoactuator Development*, General Electric Co., Schenectady, NY, USAAVLABS Technical Report 69-23 (May 1969).

[6] H. C. Kent and J. R. Sjolund, *Hydrofluidic Servoactuator Development*, Honeywell Inc., Minneapolis, MN, USAAMRDL Technical Report 73-12 (May 1973).

[7] J. R. Granan, *Research and Development on a Fluidic Servoactuator*, General Electric Co., Binghamton, NY, AFFDL-TR-70-23 (July 1970).

L_A = amplifier output inertance (Ns^2/m^5),

s = Laplace transform variable (1/s),

R_A = amplifier output resistance (Ns/m^5),

Q_i = flow into bellows (m^3/s).

The sum of the flows into the bellows is

$$Q_i - Q_o = CsP_B \; , \tag{2}$$

where

Q_o = $r_B A_B s\theta$, outflow caused by extension of bellows (m^3/s),

r_B = bellows moment arm (m),

A_B = bellows cross-sectional area (m^2),

θ = angular deflection of flapper or torque arm (rad),

C = fixed volume capacitance of bellows (m^5/N).

Substituting equation (2) into equation (1) for both sides (fig. 3 shows a push-pull system), noting that outflow on one side is inflow to the opposite side, gives the differential bellows pressure:

$$\Delta P_B(s) = \frac{\Delta P_A}{L_A C s^2 + R_A C s + 1} - \frac{2(r_B A_B)(L_A s + R_A)s\theta(s)}{L_A C s^2 + R_A C s + 1} \; . \tag{3}$$

From figure 4, the fluidic input torque, T_I (Nm), is equal to the restoring torque as shown below:

$$T_I = \Delta P_B(s)\left(r_B A_B\right) = \left(Js^2 + K_{an} + 2k_B r_B^2\right)\theta(s) \; , \tag{4}$$

where

J = polar moment of inertial of flapper ($m\text{-}N\text{-}s^2$),

K_{an} = net torque spring rate due to torsional spring, magnetic effects, and flow forces on flapper (m-N/rad),

k_B = spring rate of bellows (N/m).

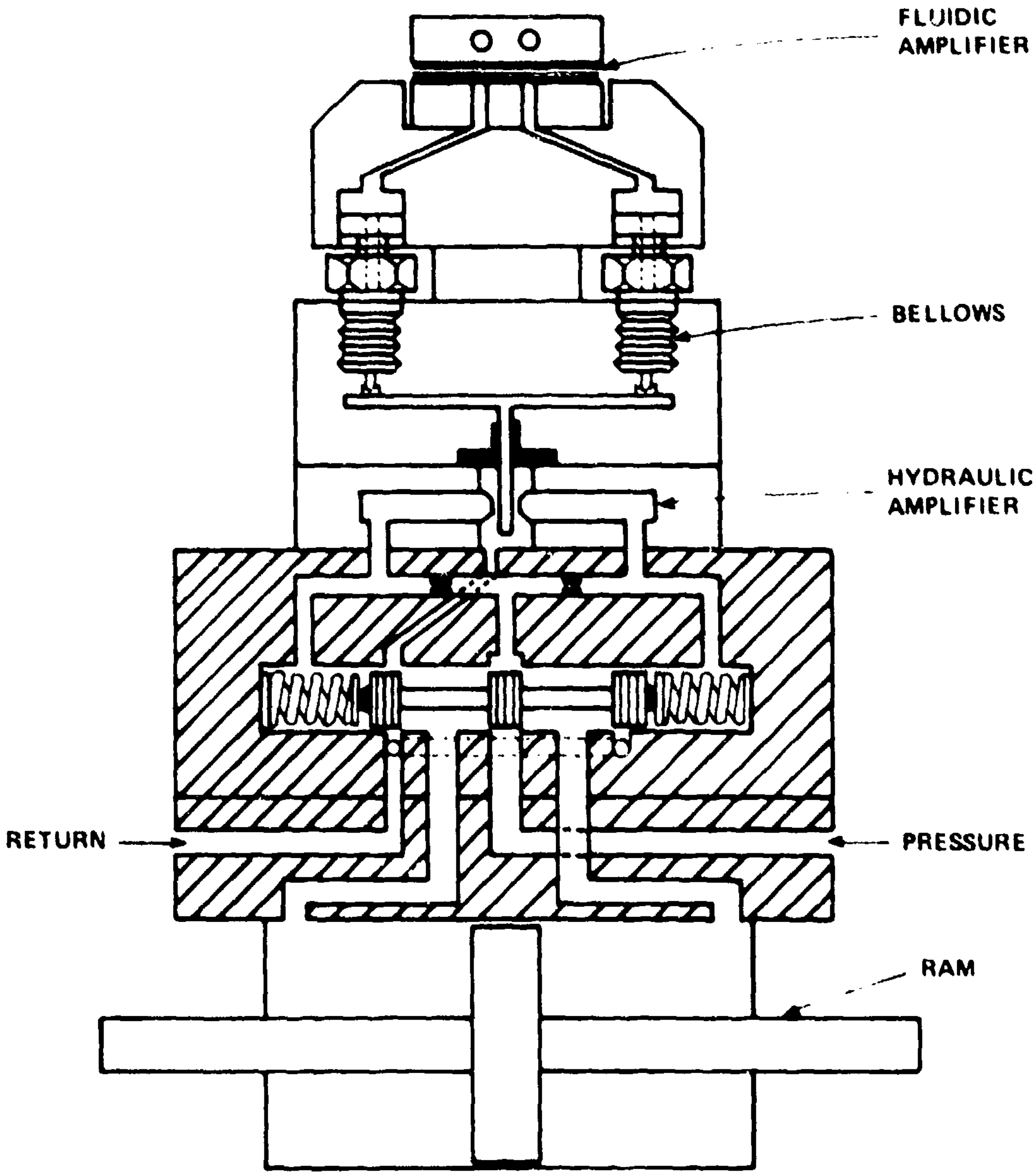

Figure 2. Two-stage fluidic input servovalve (schematic from D. Lee and D. N. Wormley, Massachusetts Institute of Technology HDL-CR-77-191-1, December 1977).

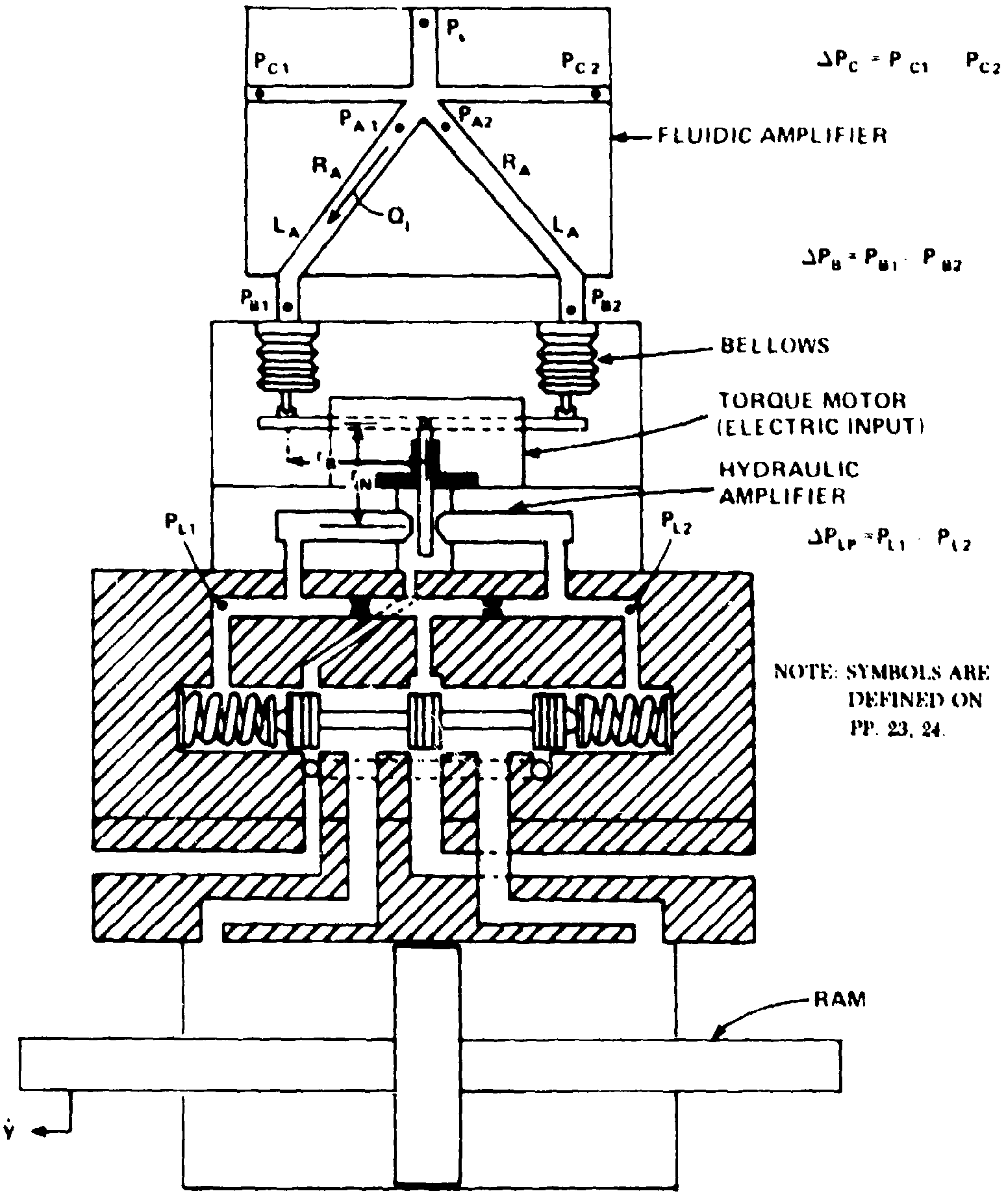

Figure 3. Fluidic input servovalve (schematic from D. Lee and D. N. Wormley, Massachusetts Institute of Technology HDL-CR-77-191-1, December 1977).

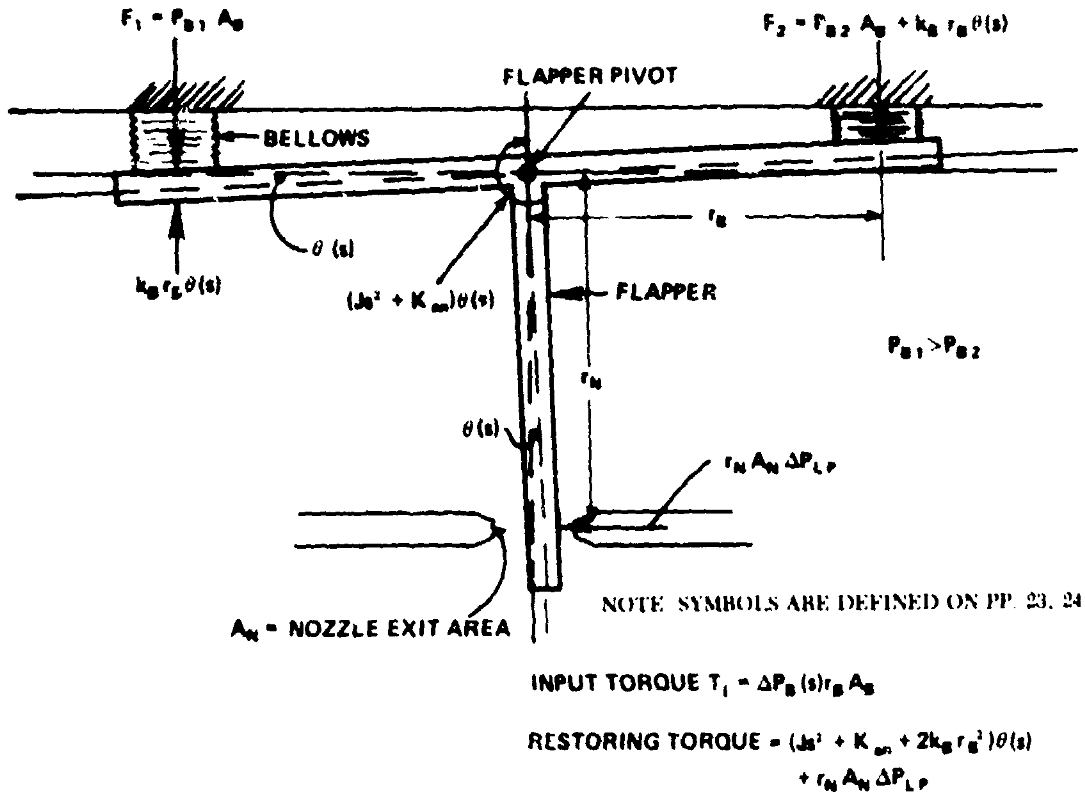

Figure 4. Torques acting on flapper.

If the flapper arm bends as shown in figure 5, the torque summation on the flapper must include the bending term, $K_{cf} r_N \phi(s)$, as shown below:

$$\Delta P_B(s)\left(r_B A_B\right) = \left(Js^2 + K_{an} 2k_B r_B^2\right)[\theta(s)] + K_{cf} r_N \phi(s) \ . \qquad (5)$$

where

K_{cf} = spring constant of cantilevered flapper arm, which is assumed to be fixed at pivot (N/rad),

r_N = flapper moment arm (m),

$\phi(s)$ = single side flapper bending angle (rad).

Substituting equation (3) into equation (5) gives the flapper deflection, $\theta(s)$ (rad),

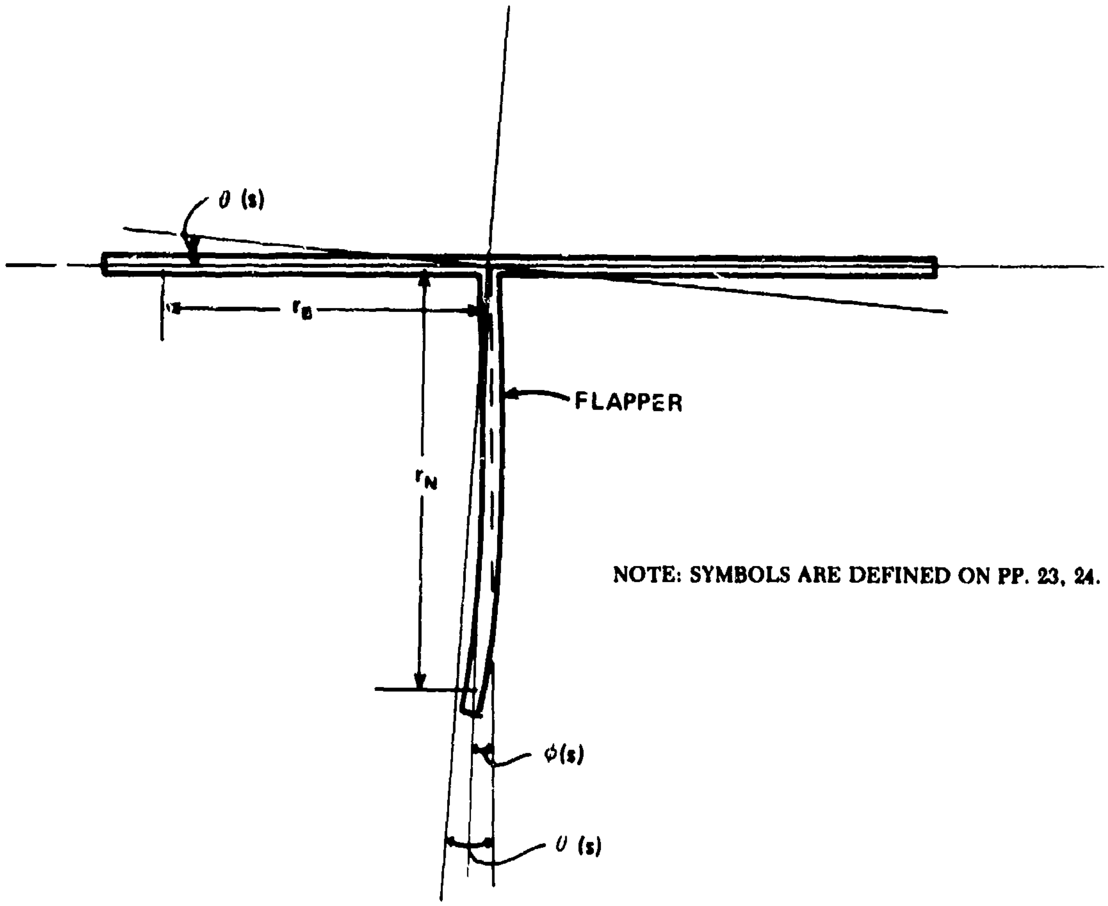

Figure 5. Flapper deflection and bending angle.

$$\theta(s) = \frac{\Delta P_A\left(r_B A_B\right) \dfrac{1}{L_A C s^2 + R_A C s + 1} - K_{cf} r_B \phi(s)}{J s^2 + \left[\dfrac{2\left(r_B A_B\right)^2\left(L_A s + R_A\right)}{L_A C s^2 + R_A C s + 1} \right] s + K_{an} + 2 k_B r_B^2} \; . \qquad (6)$$

However, if the flapper arm cantilever spring constant, K_{cf}, is very high, then $\phi(s)$ is negligible, and equation (6) reduces to

$$\theta(s) = \frac{\Delta P_A\left(r_B A_B\right) \dfrac{1}{L_A C s^2 + R_A C s + 1}}{J s^2 + \left[\dfrac{2\left(r_B A_B\right)^2\left(L_A s + R_A\right)}{L_A C s^2 + R_A C s + 1} \right] s + K_{an} + 2 k_B r_B^2} \; . \qquad (7)$$

For hydraulic applications, C = (volume/bulk modulus) << 1 and equation (7) becomes

$$\theta(s) = \frac{\Delta P_A(r_B A_B)}{\left[J + 2(r_B A_B)^2 L_A\right]s^2 + \left[2(r_B A_B)^2 R_A\right]s + K_{an} + 2k_B r_B^2} . \tag{8}$$

The flapper displacement, x_f (rad), at the nozzles is given as

$$x_f = r_N \theta . \tag{9}$$

When the flapper is deflected from its centered position, a differential pressure, ΔP_{LP} (Pa), is generated at the ends of the spool:

$$\Delta P_{LP} = K_f x_f , \tag{10}$$

where

K_f = flapper nozzle pressure gain (N/m^3).

The differential pressure acts against the centering springs at the ends of the spool. Neglecting sliding friction, the spool displacement, x_s (m), as a function of ΔP_{LP} is given as

$$x_s = \Delta P_{LP} A_s / K_s , \tag{11}$$

where

A_s = spool end area (m^2),

K_s = differential spring constant of centering springs attached to ends of spool (N/m).

Finally, the spool displacement generates a load flow, Q_L (m^3/s), given by

$$Q_L = K_{sp} x_s , \tag{12}$$

where

K_{sp} = spool flow constant (m^2/s).

A block diagram description of the complete two-stage servovalve is shown in figure 6. The spool position feedback term, $r_N A_N \Delta P_{LP}$, is negligible so that this is essentially an open loop type of servovalve.[1] The complete valve transfer function is given as

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[1] A. C. Morse, *Electrohydraulic Servomechanisms*, McGraw-Hill Book Co., New York (1963).

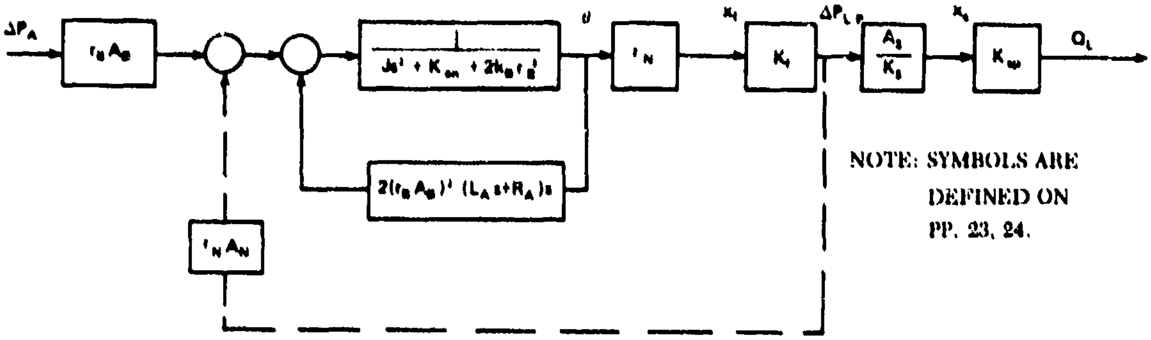

Figure 6. Fluidic input servovalve block diagram.

$$\frac{Q_L(s)}{\Delta P_A(s)} = \frac{\dfrac{r_B A_B r_N K_f A_s K_{sp}}{K_s}}{\left[J + 2(r_B A_B)^2 L_A\right]s^2 + \left[2(r_B A_B)^2 R_A\right]s + K_{an} + 2k_B r_B^2} . \tag{13}$$

Two-stage electrohydraulic servovalves are complex devices that exhibit high-order, nonlinear response. If a first-, second-, or third-order transfer function, H(s) (m^3/Ns) is selected to represent servovalve dynamics, only an approximation to the actual response is possible. First-order approximations result in the expression "equivalent time constant" of the servovalve, τ. This approximation assumes that the servovalve can be described as a first-order system given by equation (14):

$$H_1(s) = \frac{K_1}{\tau s + 1} , \tag{14}$$

where

K_1 = servovalve gain (m^3/Ns).

This approximation should correspond to the 45-deg phase point rather than the 0.7 amplitude point (-3 dB). This representation of the servovalve dynamics is good through the low-frequency range, approximately 0 to 50 Hz.* If the low-frequency range of the fluidic input valve is of interest, then the s^2 term in equation (13) can be neglected compared with the (K_{an} + $2k_B r_B^2$) term. This approximation implies that the fluidic amplifier output inertance, L_A, and the flapper polar moment of inertia, J, are negligible in this frequency range. The valve transfer function then becomes

*D. J. Thayer, *Transfer Functions for Moog Servovalves, Rev. ed., Moog Inc., East Aurora, NY, Technical Bulletin 103* (1965).

$$H_1(s) = \frac{Q_L(s)}{\Delta P_A(s)} = \frac{\dfrac{A_B K_f A_s K_{sp}}{\left(\dfrac{K_{an}}{r_B^2} + 2k_B\right) K_s}}{\left(\dfrac{2A_B^2}{\dfrac{K_{an}}{r_B^2} + 2k_B}\right) R_A s + 1} , \quad (15)$$

where

$r_B \cong r_N$.

Thus, the servovalve time constant, τ, is given as

$$\tau = \left(\frac{2A_B^2}{\dfrac{K_{an}}{r_B^2} + 2k_B}\right) R_A , \quad (16)$$

where

$2A_B^2/[(K_{an}/r_B^2) + 2k_B]$ = effective capacitance of bellows.

A phase lag of 45 deg occurs at the first-order break frequency, f_B (1/s), given by

$$f_B = \frac{1}{2\pi\tau} . \quad (17)$$

Therefore, τ should be minimized to achieve favorable high-frequency response. From equation (16), the design parameters affecting τ are A_B, K_{an}, r_B, k_B, and R_A. R_A is determined by the available system flow, and K_{an} is fixed by the flapper nozzle and the second-stage spool design. Parameters A_B, r_B, and k_B can all be used to minimize τ. Decreasing r_B decreases τ; however, it decreases also the servovalve gain $\{A_B K_f A_s K_{sp}/[(K_{an}/r_B^2) + 2k_B]K_s\}$ from equation (15) by the same magnitude. Increasing k_B decreases τ, but it decreases also the servovalve gain by the same magnitude. Decreasing the bellows area decreases τ by the area squared, A_B^2. It decreases also the servovalve gain, but to the first power, A_B. Since usually $A_B < 1$, then for any decrease in A_B, the decrease is greater in τ, which has the factor A_B^2, than in the servovalve gain, which has the factor A_B. Therefore, as a first step, the bellows area should be reduced to minimize τ.

A second-order approximation to servovalve dynamics is used when response near the 90-deg phase lag point is of interest. This approximation is usually used in describing position control servomechanisms, that is, closed-loop position control systems. The 90-deg phase lag point is best associated with the apparent natural frequency (or natural frequency), ω_n (1/s), of the servovalve, and the damping ratio, ξ, is best associated with the amplitude characteristic.* The second-order approximation to servovalve dynamics has the form

*D. J. Thayer, *Transfer Functions for Moog Servovalves*, Rev. ed., Moog Inc., East Aurora, NY, Technical Bulletin 103 (1965).

$$H_2(s) = \frac{\left(\frac{1}{\omega_n}\right)^2 K_2}{\left(\frac{s}{\omega_n}\right)^2 + \left(\frac{2\xi}{\omega_n}\right)s + 1}\,, \tag{18}$$

where

K_2 = servovalve gain (m^5/Ns^3).

If the frequency response near the 90-deg phase lag point is of interest for the fluidic input servovalve, then equation (13) can be rewritten to approximate the servovalve transfer function, $Q_L(s)/\Delta P_A(s)$, as

$$\frac{Q_L(s)}{\Delta P_A(s)} = \frac{\dfrac{r_B^2 A_B K_f A_s K_{sp}}{(K_{an} + 2k_B r_B^2) K_s}}{\left[\dfrac{J + 2(r_B A_B)^2 L_A}{K_{an} + 2k_B r_B^2}\right] s^2 + \left[\dfrac{2(r_B A_B)^2 R_A}{K_{an} + 2k_B r_B^2}\right] s + 1}\,, \tag{19}$$

where

$$\omega_n^2 = \frac{K_{an} + 2k_B r_B^2}{J + 2(r_B A_B)^2 L_A}\,,$$

$$K_2 = \frac{\dfrac{r_B^2 A_B K_f A_s K_{sp}}{K_s}}{J + 2(r_B A_B)^2 L_A}\,,$$

$$\frac{2\xi}{\omega_n} = \frac{2(r_B A_B)^2 R_A}{K_{an} + 2k_B r_B^2}\,,$$

$$\xi = \frac{(r_B A_B)^2 R_A}{\{(K_{an} + 2k_B r_B^2)[J + 2(r_B A_B)^2 L_A]\}^{1/2}}\,.$$

A better second-order approximation of servovalve response requires that the input dynamics and the transport delay of the fluidic amplifier be included in equation (19). A high-performance servovalve calls for the natural frequency to be as large as possible and the damping ratio to be in the range $0.7 \leq \zeta \leq 1.0$. From equation (19), a high natural frequency requires that the bellows area be as small as possible and, for a given bellows area, that the amplifier output resistance be adjusted so that ζ is in the desired range. Good first- or second-order servovalve dynamics call for the bellows area to be as small as possible.

2.2 Fluidic Amplifier Leakage Flow

The fluidic amplifier portion of the fluidic input servovalve (fig. 2) is part of the first stage of the valve. Therefore, the fluidic amplifier leakage flow adds to the existing first-stage leakage flow of the flapper nozzle valve. The amplifier leakage flow is the amplifier supply flow, Q_s (m^3/s). The necessary supply flow is a function of the amplifier output resistance and the desired servovalve time constant. The supply flow is derived in terms of these parameters.

The amplifier supply flow can be written as

$$Q_s = \frac{P_s}{R_s} = \frac{a\Delta P_A}{nR_A} , \tag{20}$$

where

P_s = amplifier supply pressure (Pa),

R_s = amplifier power nozzle resistance (Ns/m^5),

a = constant (with values in range $0.5 \leq a \leq 0.6$),

n = constant (determined by amplifier height and number of parallel laminates used).

From equation (8), the maximum flapper deflection, θ_{max} (rad), occurs when $s \to 0$ and is given as

$$\theta_{max} = \frac{r_B A_B \Delta P_A}{K_{an} + 2k_B r_B^2} . \tag{21}$$

From equation (21), ΔP_A can be written as

$$\Delta P_A = \frac{\left[\frac{K_{an}}{r_B^2} + 2k_B\right] r_B \theta_{max}}{A_B} . \tag{22}$$

If a first-order approximation is used to describe the servovalve dynamics, the time constant from equation (16) is

$$\tau = \frac{2A_B^2 R_A}{\dfrac{K_{an}}{r_B^2} + 2k_B} .$$

From equation (16), R_A can be written as

$$R_A = \frac{\left[\dfrac{K_{an}}{r_B^2} + 2k_B\right]\tau}{2A_B^2} , \tag{23}$$

and, by using equations (22) and (23), the supply flow can be written as

$$Q_s = \frac{2ar_B\theta_{max}A_B}{n\tau} . \tag{24}$$

Thus, minimizing the bellows area minimizes also the amplifier supply flow or the first-stage servovalve leakage flow.

3. TEST RESULTS

The Harry Diamond Laboratories (HDL) purchased two fluidic input servovalves with essentially identical performance specifications (fig. 2). Cursory tests were conducted on one servovalve at HDL. Since HDL does not have the facilities for thoroughly testing servovalves, the other one was further developed and thoroughly tested on contract.[9] The additional development allowed the servovalve to be driven by an electrical signal or a fluidic signal. The electrical signal energized the torque motor, which in turn drove the flapper nozzle valve. The fluidic signal was amplified by a fluidic amplifier, which drove a set of bellows, which in turn drove the flapper nozzle valve. This servovalve had an initial set of bellows with an area smaller than normal, $A_B = 31.9$ mm^2. The test program called for replacing this set of bellows with a smaller set, $A_B' = 18.1$ mm^2, and then with a larger set, $A_B'' = 44.5$ mm^2. This procedure was set up as a means of verifying the conclusions reached in the design considerations (sect. 2). The servovalve was tested with the initial set of bellows, $A_B = 31.9$ mm^2. Figure 7 shows the dynamic response of the servovalve driven by an electrical signal and by a fluidic signal.[9] The curve of phase lag versus frequency for the fluidically driven servovalve shows 45 deg of phase lag at 20 Hz. From equation (16), the servovalve time constant can be given as

$$\tau = \frac{1}{\omega}\tan\theta = \frac{1}{2\pi f}\tan 45 = 8 \text{ ms} , \tag{25}$$

[9] *D. Lee and D. N. Wormley, Hydraulic Signal-Processing Amplifier Performance in Position Control Systems, Massachusetts Institute of Technology, Cambridge, MA, HDL-CR-77-191-1 (December 1977).*

where

$$\theta = 45 \text{ deg},$$

$$f = 20 \text{ Hz}.$$

This servovalve response to fluidic input represents a significant improvement over previous fluidic input valve responses. This improvement was accomplished by using (1) smaller bellows and (2) a fluidic amplifier with little low-frequency phase shift. The initial set of bellows, A_B, was not replaced after the testing reported by Lee and Wormley,* that is, with the smaller set of bellows, A'_B, or the larger set of bellows, A''_B, because the servovalve response with the initial set of bellows was close to the desired response and because the bellows were a cost and reliability problem. For a normal production run of bellows, the spring rates vary ±30 percent from the nominal value. To obtain two bellows with identical spring rates increases the cost considerably. Moreover, the bellows ruptured very easily due to either overpressuring or mishandling.

Even though the servovalve response to fluidic input represents a significant improvement, the response is not comparable to the response to electrical input as seen in figure 7. The added phase shift using fluidic input is due to phase lag in the fluidic amplifier and the amplifier output

NOTE: SYMBOLS ARE DEFINED ON PP. 23, 24.

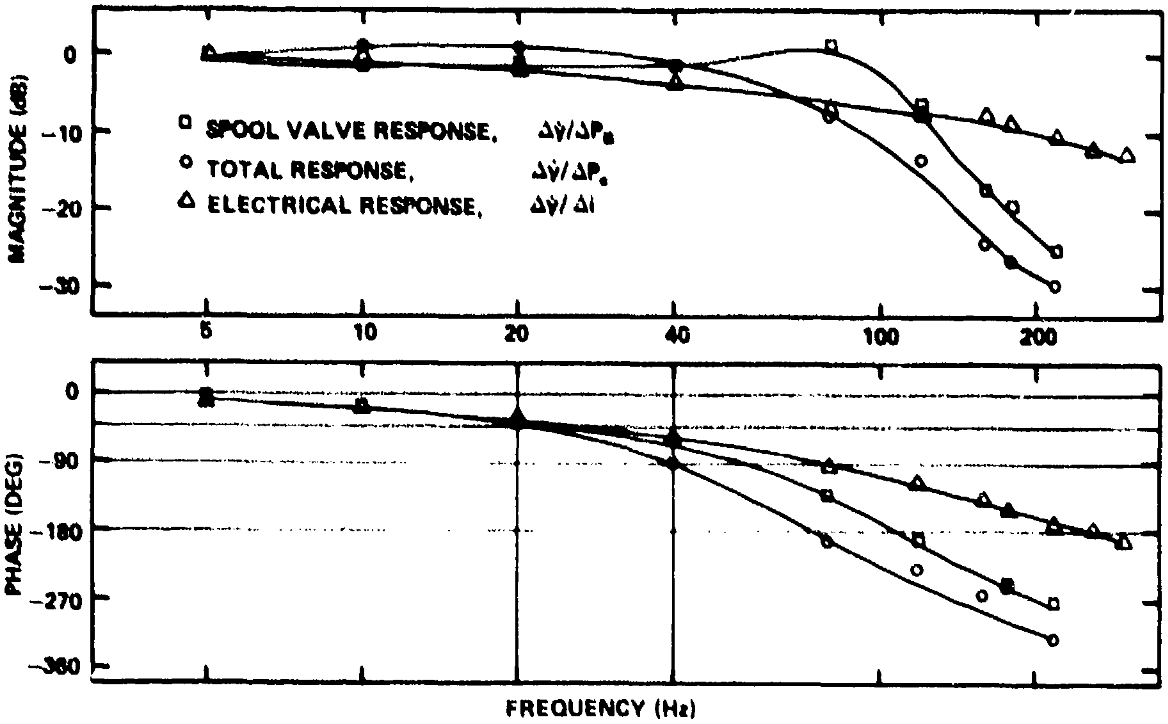

Figure 7. Fluidic input servovalve (fig. 2) frequency response (data from D. Lee and D. N. Wormley, Massachusetts Institute of Technology HDL-CR-77-191-1, December 1977).

*D. Lee and D. N. Wormley, *Hydraulic Signal-Processing Amplifier Performance in Position Control Systems*, Massachusetts Institute of Technology, Cambridge, MA, HDL-CR-77-191-1 (December 1977).

resistance and bellows capacitance time constant. The phase shift observed with the fluidic input valve due to the time constant is described by equation (15), and the added phase shift due to the fluidic amplifier dynamics can be described by equation (13) with the addition of the amplifier input dynamics and transport delay. The added phase shift observed using fluidic input to the servovalve in the 20- to 120-Hz region became significant when the valve was tested* in a closed-loop servo control system. These results indicate that servovalve response to fluidic input cannot be meaningfully approximated as a first-order system (sect. 2). The frequency response of the amplifier must be considered. The results indicate also that further development is needed to make the valve response to fluidic input comparable to valve response to electrical input, possibly by using an approach that does not require bellows. This development is crucial because, at present, fluidic control system performance is degraded by the servovalve, and this degradation is independent of any fluidic sensing, amplification, and signal processing errors.

4. CONCLUSIONS

This design study is concerned with two-stage fluidic input servovalves using bellows to drive a first-stage flapper nozzle valve. The study shows that minimizing the bellows area reduces (1) the servovalve time constant and (2) the first-stage fluidic amplifier leakage flow. A reduced fluidic input servovalve time constant in the 1- to 5-ms range is desired. Experimental tests were conducted to verify the results of the design study. These tests of a dual-input (electrical and fluidic) servovalve conducted at the Massachusetts Institute of Technology show that a time constant, τ = 8 ms, was obtained by using very small bellows. The first-order approximation to a servovalve response used in the design consideration was found to be valid up to 20 Hz or for a phase lag to 45 deg. This servovalve response to fluidic input represents a significant improvement over previous fluidic input servovalve response. However, the response of the servovalve to fluidic input is not comparable to the response to electrical input beyond 20 Hz. The two servovalve responses differ in that there is additional phase lag by using fluidic input. The added phase shift was due to (1) the fluidic amplifier phase lag and (2) the amplifier output resistance and bellows capacitance time constant. This added phase lag degraded the system performance when the servovalve was used with fluidic input in a closed-loop servo system. The experimental results indicate that servovalve response to fluidic input must be approximated as a second- or higher-order system when used in a closed-loop servo system. The approximate response must take into account (1) the phase shift of the fluidic amplifier and (2) the amplifier output resistance and bellows capacitance time constant. The results indicate also that further development is needed to make the fluidic input valve response comparable to the electrical input valve response.

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*D. Lee and D. N. Wormley, *Hydraulic Signal-Processing Amplifier Performance in Position Control Systems*, Massachusetts Institute of Technology, Cambridge, MA, HDL-CR-77-191-1 (December 1977).

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NOMENCLATURE

a	Constant (with values in range $0.5 < a < 0.6$)
A_B	Bellows cross-sectional area (m^2)
A_s	Spool end area (m^2)
A_N	Nozzle exit area (m^2)
C	Fixed volume capacitance of bellows (m^5/N)
f	Frequency (Hz)
f_B	Break frequency (1/s)
$H(s)$	Transfer function (m^5/Ns)
$H_1(s)$	Servovalve transfer function, first-order approximation (m^5/Ns)
$H_2(s)$	Servovalve transfer function, second-order approximation (m^5/Ns)
J	Polar moment of inertia of flapper (m-N-s^2)
k_B	Spring rate of bellows (N/m)
K_{an}	Net torque spring rate due to torsional spring, magnetic effects, and flow forces on flapper (m-N/rad)
K_{cf}	Spring constant of cantilevered flapper arm, which is assumed to be fixed at pivot (N/rad)
K_f	Flapper nozzle pressure gain (N/m^3)
K_s	Differential spring constant of centering springs attached to ends of spool (N/m)
K_{sp}	Spool flow constant (m^2/s)
K_1	Servovalve gain, first-order approximation (m^5/Ns)
K_2	Servovalve gain, second-order approximation (m^5/Ns^3)
L_A	Amplifier output inertance (Ns^2/m^5)
n	Constant (determined by amplifier height and number of parallel laminates used)
P_A	Amplifier output pressure (Pa)
P_B	Bellows pressure (Pa)
P_c	Amplifier control pressure (Pa)
P_{LP}	Differential pressure (Pa)
P_s	Amplifier supply pressure (Pa)
Q_i	Flow into bellows (m^3/s)
Q_L	Load flow (m^3/s)
Q_o	Flow out of bellows (m^3/s)
Q_s	Amplifier supply flow (m^3/s)
r_B	Bellows moment arm (m)
r_N	Flapper moment arm (m)
R_A	Amplifier output resistance (Ns/m^5)
R_s	Amplifier power nozzle resistance (Ns/m^5)

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NOMENCLATURE (Cont'd)

s	Laplace transform variable (1/s)
T_I	Fluidic input torque (Nm)
x_f	Flapper displacement (m)
x_s	Spool displacement (m)
$\dot{y}$	Load velocity (m/s)
θ	Angular deflection of flapper or torque arm (rad)
θ_{max}	Maximum flapper deflection (rad)
$\theta(s)$	Flapper deflection (rad)
ξ	Damping ratio
τ	Equivalent servovalve time constant (s)
$\phi(s)$	Single side flapper bending angle (rad)
ω_n	Apparent natural frequency or natural frequency (1/s)

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